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Experimental Investigation of Convective Heat Transfer and Pressure Drop of Al^2O^3/Water Nanofluid in Laminar Flow Regime inside a Circular Tube

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***Abstract*—**In the present study, Convective heat transfer coefficient and pressure drop of Al_2O_3/water nanofluid in laminar flow regime under constant heat flux conditions inside a circular tube were experimentally investigated. Al_2O_3/water nanofluid with 0.5% and 1% volume concentrations with 15 nm diameter nanoparticles were used as working fluid. The effect of different volume concentrations on convective heat transfer coefficient and friction factor was studied. The results emphasize that increasing of particle volume concentration leads to enhance convective heat transfer coefficient. Measurements show the average heat transfer coefficient enhanced about 11-20% with 0.5% volume concentration and increased about 16-27% with 1% volume concentration compared to distilled water. In addition, the convective heat transfer coefficient of nanofluid enhances with increase in heat flux. From the results, the average ratio of (f_{nf}/f_{bf}) was about 1.10 for 0.5% volume concentration. Therefore, there is no significant increase in friction factor for nanofluids.

***Keywords*—**Convective heat transfer, Laminar flow regime, Nanofluids, Pressure drop

I. Introduction

IN the last decades, many efforts have been made to produce more efficient heat exchangers in order to conserve energy. These efforts deal with improving the heat transfer rate by means of extended surfaces, mini-channels and micro-channels. In addition, many studies have been carried out on thermal properties of suspensions of solid particles in conventional heat transfer fluids to enhance their poor thermal performance.

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For this purpose, Ahuja [1] dispersed micro/millimeter sized particles with high thermal conductivity in the base fluid. Choi et al. [2] dispersed nanometer sized particles in the base fluid which were called nanofluid. Nanofluids were an improvement on previous suspensions, because they suffered from less sedimentation, large pressure drops, and were less corrosive. The development of nanofluids as a new class of heat transfer with the use of nanotechnology has been the subject of much attention in recent years. Hwang et al. [3] investigated the convective heat transfer coefficient and pressure drop of Al_2O_3/water nanofluids inside a circular tube in the fully developed laminar flow regime with constant heat flux. They reported that, convective heat transfer coefficient increased by up to 8% at 0.3% volume concentration. Williams et al. [4] studied the turbulent convective heat transfer coefficient of alumina and zirconia nanofluids in a horizontal tube. They observed no significant heat transfer enhancement for nanofluid compared to base fluid. Also, they demonstrated that the convective heat transfer coefficient could be predicted by means of the traditional correlations. Maiga et al. [5], [6] numerically studied the hydrodynamic and thermal characteristics of forced convection flow of nanofluids inside a tube with constant heat flux. Results showed that the addition of nanoparticles increased the heat transfer coefficient significantly. Convective heat transfer of different nanofluids (CuO, Al_2O_3 and SiO_2) in an ethylene glycol and water mixture, flowing inside a circular tube under constant heat flux condition under turbulent regime have been numerically analyzed by Namburu et al. [7] They found that, CuO nanofluids showed higher heat transfer performance at the same volume concentration and specific Reynolds number. Lee et al. [8] experimentally investigated the thermal conductivity of nanofluids. They measured the thermal conductivity of four different nanofluids (Al_2O_3/water, Al_2O_3/ethylene glycol, CuO/water, and CuO/ethylene glycol) by a transient hot wire method. They showed the conductivity ratio enhancement of ethylene glycol based nanofluid was always higher than those of water based nanofluids. In addition, the thermal conductivity of nanofluid depends on particles shape and size, and also the thermal properties of base fluid and nanoparticles.

Das et al. [9] examined the effect of temperature on thermal conductivity of two different nanofluids (Al_2O_3/water or CuO/water). The results showed that the More the temperature is more the thermal conductivity is, and the effective thermal conductivity models failed to predict the thermal conductivity of nanofluids. Heris et al. [10], [11] studied the heat transfer characteristics of alumina and copper oxide nanofluids under laminar flow regime inside a circular tube with constant wall temperature boundary condition. They observed higher enhancement in alumina nanofluid than copper oxide and reported heat transfer coefficient enhancement for both nanofluids with increasing nanoparticle concentrations. Considering that thermo physical properties of nanofluids greatly depend on the type and physical properties of nanoparticles being used, Therefore there is a widespread need for more studies and experimental results for heat transfer properties of nanofluids. In the present paper, convective heat transfer coefficient and pressure drop of Al_2O_3/water nanofluid in laminar flow regime under constant heat flux conditions inside a circular tube were experimentally investigated in order to present a deep insight into the current issue.

II. Preparation of Nanofluid

Al_2O_3 nanoparticles will be used here (from TECNAN Company)[12]. Table I presents the physical properties of the particles.

TABLE I
PROPERTIES OF THE AL_2O_3 NANOPARTICLES

CHEMICAL FORMULA	γ- Al2O3
COLOUR	White
MORPHOLOGY	Spherical
CRYSTAL PHASE	Gamma
SPECIFIC SURFACE AREA (SSA)	90-160 m2/g
TRUE DENSITY	3.65 g/cc
PORE VOLUME	0,391 cc/g
AVERAGE PORE SIZE	133,64 Å

Fig. 2 shows the TEM image provided by the research support center of the University of Alconte, using a JEOL, MOD.JEM2010. Fig.1 presents The XRD image provided by a Bruke D-8 Advance. It indicates the particle diameter to be in the range of 10-20 nm. Hence, we consider the average particle size to be 15 nm.

Al2O3/water nanofluids with 0.5% and 1% volume fractions were provided using distilled water as base fluid. The nanofluids were stabilized by using a ultrasonic cleaner (CD-4820 with 170W and 50Hz) and an electromagnetic stirrer for about 4 hours. Through the experiments nanofluid was found to remain stable.

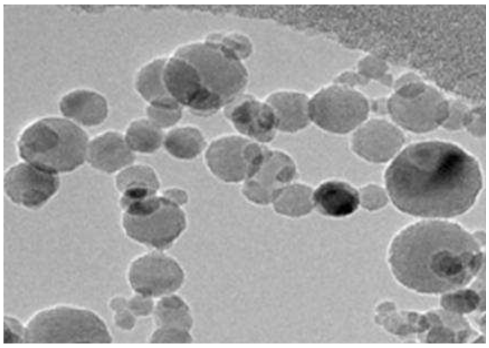

Fig. 1 TEM image of the Al_2O_3 nanoparticles

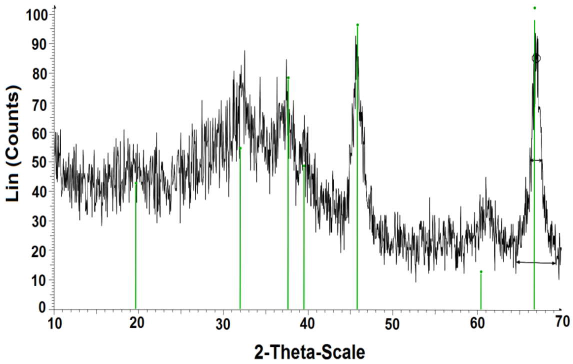

Fig. 2 XRD image of the Al_2O_3 nanoparticles

III. Experimental Setup

A schematic of the experimental setup used for measuring convective heat transfer and pressure drop is shown in Fig. 3. It includes a pump, calming section, test section, cooling unit, riser section secondary and main reservoir. A three litter glass reservoir covered with glass wool is used as the main reservoir. The working fluid is pumped using (MULTI 5800 SICCE). At the pump outlet a bypass valve is installed to ensure better flow rate control. In order to ensure a fully developed laminar flow, a calming section is used. The test section consists of a circular copper tube of 7 mm ID and 9.46mm OD and 1000mm length. Nickel chrome wire is wound around the test section tube and a glass wool cover is added to avoid any radial heat loss. The two ends of the test section are connected to rest of circulatory system using plastic flanges to prevent any axial heat transfer. The terminals of the nickel chrome wire are attached to an AC power supply. Therefore, the heat flux can be adjusted by varying the voltage. Nine SMT-160 thermocouples are evenly placed on the test section to measure wall temperature. Two thermocouples are placed along the flow line to measure inlet and outlet temperatures. Data are collected from the thermocouples every 5 seconds by computer. Two manometers are used to measure the pressure drops along the test section. After passing through the test section, the fluid flows through the riser section to ensure continuity and then the flow rate is measured by collecting the fluid for a period of time with the help of a precise measuring jar and a stop watch.

After that, the fluid enters the cooling unit which is a counter flow double pipe heat exchanger with a cold water supply and a pump for cold water circulation inside the heat exchanger. A three way valve is placed before the main reservoir for cleaning the system.

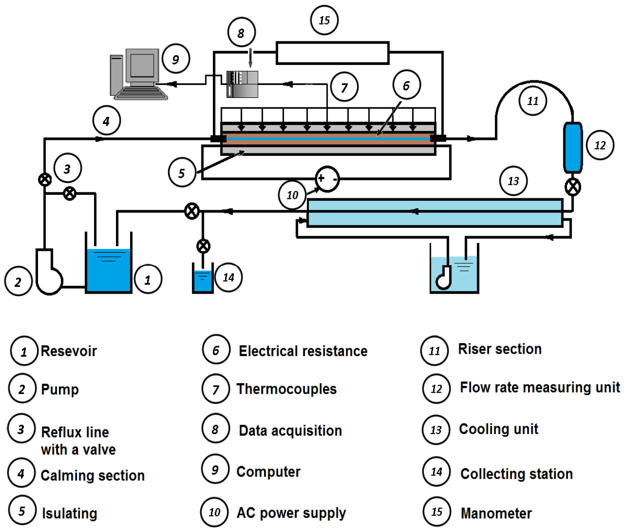

Fig. 3 Schematic view of the experimental setup

IV. Data and Processing

A. Thermo Physical Properties of Nanofluids

All thermophysical properties of nanofluid are calculated with equations (1) to (4) which can be found in the literature [13]-[16]. All properties are calculated using bulk temperatures between inlet and outlet.

$$\rho_{nf} = (1-\varphi)\rho_{bf} + \varphi\rho_p \tag{1}$$

$$(\rho C_p)_{nf} = (1-\varphi)(\rho C_p)_{bf} + \varphi(\rho C_p)_p \tag{2}$$

$$\mu_{nf} = (1+2.5\varphi)\mu_{bf} \tag{3}$$

Where φ is the volume concentration and μ is the dynamic viscosity. The index *nf*, *bf* and *p* refers to nanofluid, base fluid and particle properties respectively.

$$k_{nf} = k\left[\frac{k_p + 2k + 2(k_p - k)(1+\beta)^3\varphi}{k_p + 2k - 2(k_p - k)(1+\beta)^3\varphi}\right] \tag{4}$$

$$\mathrm{Re}_{nf} = \frac{\rho_{nf} v d}{\mu_{nf}} \tag{5}$$

$$\mathrm{Pr}_{nf} = \frac{\mu_{nf} C_{p_{nf}}}{k_{nf}} \tag{6}$$

Where β is the ratio of nanolayer thickness to the nanoparticle diameter which is considered to be 0.1 in equation (4) of this study [16]. In additions k is the thermal conductivity, *Re* is the Reynolds number and *Pr* is the Prandtl number.

B. Heat Transfer and Hydrodynamics Calculation

According to equation (7)-(8) the local Nusselt number (Nu) and heat transfer coefficient (h) are defined by:

$$Nu(x) = \frac{h(x)d}{k} \tag{7}$$

$$h(x) = \frac{q''}{(T_w - T_f)_x} \tag{8}$$

Where q'' is the heat flux, T_w and T_f are the wall and fluid temperatures respectively and x is the axial distance from test section inlet. Heat flux was calculated from below:

$$q'' = \frac{Q}{\pi d L} \tag{9}$$

Where d is the tube diameter, L the test section length and Q the heat transfer rate which is obtained by equation (10).

$$Q = \frac{Q_1 + Q_2}{2} \tag{10}$$

Where Q_1 is the heat transfer rate from the AC power unit and Q_2 is the heat transfer rate measured from the working fluid.

$$Q_1 = \frac{V^2}{R} - Q_{loss} \tag{11}$$

Where V is the AC power unit voltage, R is the nickel chrome wire resistance and Q_{loss} is the heat loss.

$$Q_2 = \dot{m} C_p (T_{out} - T_{in}) \tag{12}$$

Using the principle of conservation of energy, the temperature profile inside the tube can be described by:

$$T_f = T_{in} + \frac{q'' P x}{\rho C_p v A} \tag{13}$$

Where C_p is the heat capacity, ρ is the fluid mass density, A and P are the cross-sectional area and perimeter respectively of the test tube, and v is the average fluid velocity. The average convective heat transfer coefficient can be obtained from equation (14).

$$\bar{h} = \frac{q''}{(\bar{T}_w - \bar{T}_f)} \tag{14}$$

Where $\bar{T}_w$ is the average wall temperature and $\bar{T}_f$ is the average wall bulk temperature of fluid.

The pressure drop can be calculated using equation (15).

$$h_l = f\,\frac{L}{d}\,\frac{v^2}{2g} \tag{15}$$

V. Results and Discussion

A. Convective Heat Transfer Coefficient

In order to assess the accuracy of the experimental setup, the Nusselt number obtained from equation (16) [17, 18] was compared with experimental data for distilled water at

Re=483.24 and constant heat flux of *q″=10500 w/m²*. Fig. 5 shows a better agreement between the experimental results and those predicted by Shah equation.

$$Nu_{x_*} = \begin{cases} 3.303 x_*^{-1/3} - 1.00 & x_* \le 0.00005 \\ 1.302 x_*^{-1/3} - 0.50 & 0.00005 \le x_* \le 0.0015 \\ 4.364 + 8.68(10^3 x_*)^{-0.506} e^{-41 x_*} & x_* > 0.001 \end{cases} \quad (16)$$

Where x_* was calculated from below

$$x_* = \frac{x}{d \,.\, \mathrm{Re} \,.\, \mathrm{Pr}} .$$

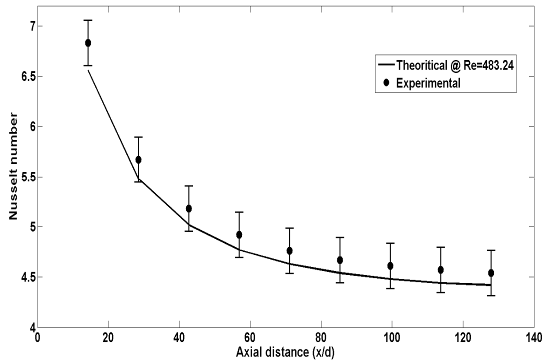

Fig. 5 Comparison of the experimental Nusselt number with Shah equation versus axial distance

Also Fig. 6 shows the ratio of experimental friction factor to equation (17) prediction for distilled water versus Re number under ambient conditions in laminar flow regime. As it is shown the experimental data had good agreement with theoretical results.

$$f = \frac{64}{\mathrm{Re}} \quad (17)$$

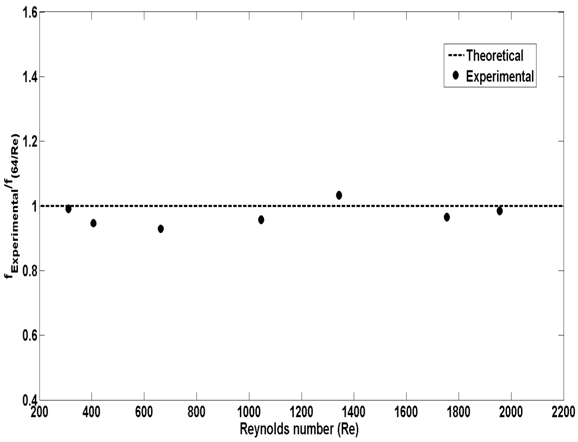

Fig. 6 Ratio of experimental friction factor to theoretical prediction for distilled water versus Re number

Fig. 7 demonstrates the convective heat transfer coefficient of nanofluid with different volume concentrations and distilled water versus Re number at axial location *x/d=71.07* and constant heat flux of *q″=10500 w/m²*. The results indicate increasing particle concentration leads to a better enhancement in convective heat transfer coefficient.

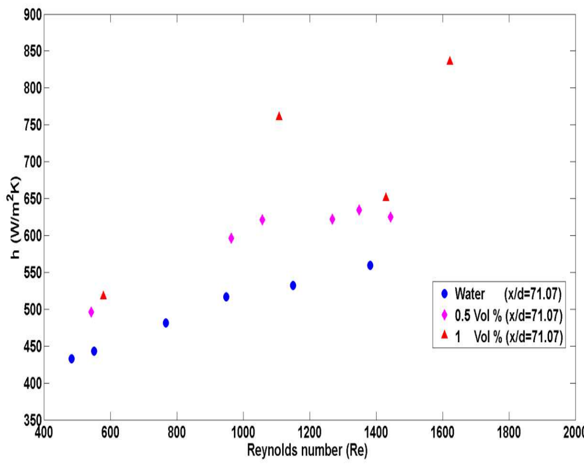

Fig. 7 Convective heat transfer coefficient of nanofluid and water versus Re number at *x/d=71.07*

Fig.8 presents the ratio of convective heat transfer coefficient of γ-Al_2O_3 nanofluid for 0.5% and 1% volume concentration to the base fluid versus Re number with constant heat flux of *q″=10500 w/m²*. From the results, (h_{nf}/h_{bf}) rises with increasing Re number for both volume concentrations. Measurement show the average heat transfer coefficient enhanced about 11-20% with 0.5% volume concentration and increased about 16-27% with 1% volume concentration compared to distilled water.

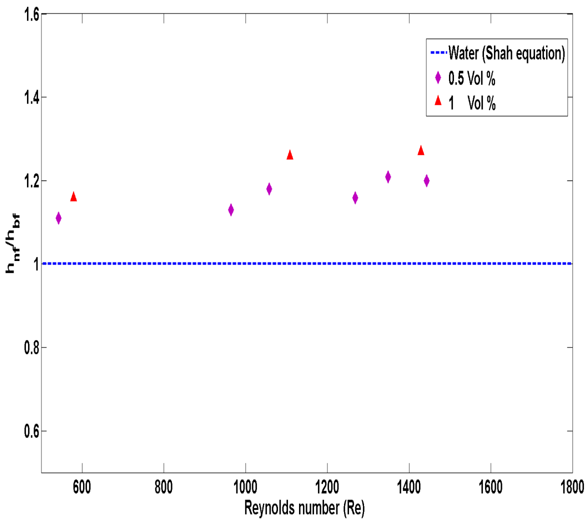

Fig. 8 Ratio of convective heat transfer coefficient of nanofluids to the base fluid versus Re number

These enhancements are caused by Brownian motion, particle migration and re-arrangement of the particles and reduction of boundary layer thickness.

Fig. 9 presents the convective heat transfer coefficient of nanofluid with different volume concentrations and distilled water versus axial distance for *Re=900* and heat flux of *q″=9000 w/m²*. The enhancement decreases with increasing axial distance. Therefore, in order to better utilize the heat transferring capabilities of nanofluids, it is better to employ them in heat exchanging equipment with low x/d ratios.

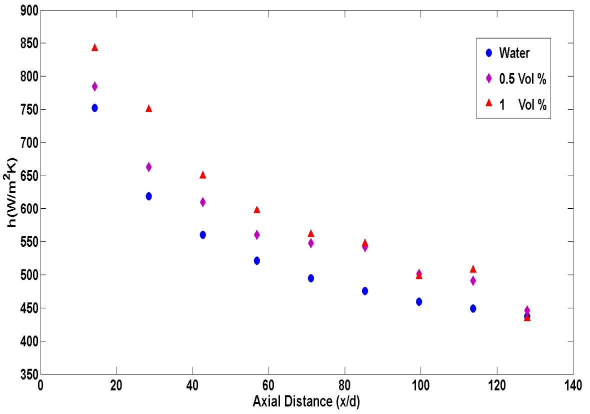

Fig. 9 Convective heat transfer coefficient of nanofluid versus axial distance for *Re=900*

The effect of various heat fluxes on convective heat transfer coefficient of nanofluid with 1% volume concentration at different Reynolds numbers was investigated. As Fig. 10 indicates, the convective heat transfer coefficient of nanofluid enhances with increase in heat flux. This enhancement is more evident at higher Reynolds numbers.

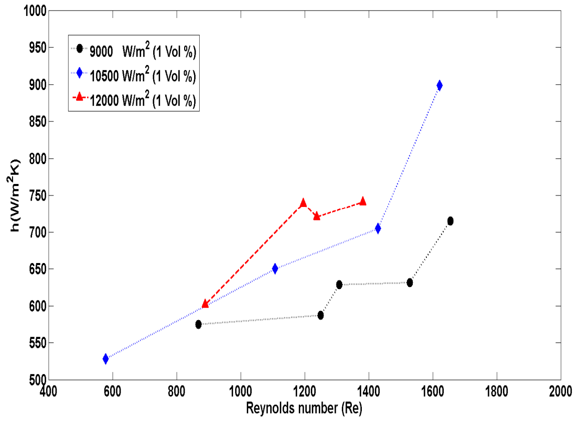

Fig. 10 Convective heat transfer coefficient of nanofluids versus Re number for various heat fluxes

B. Pressure Drop

In Fig. 11, the ratio of friction factor of γ- Al_2O_3 nanofluid to the base fluid versus Re number was measured. Results show the average ratio of (f_{nf}/f_{bf}) was about 1.10 for 0.5% volume concentration. Therefore, there is no significant increase in friction factor for nanofluids.

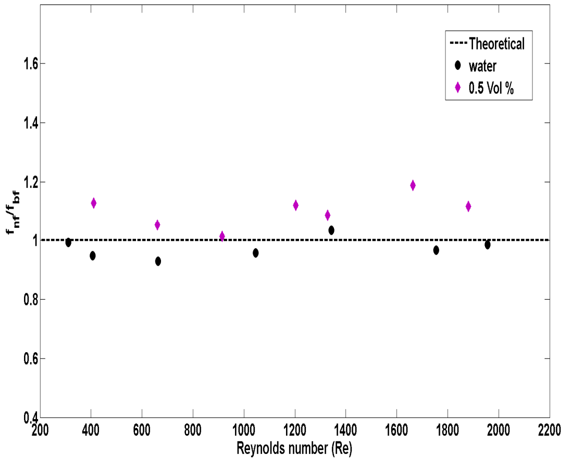

Fig. 11 Ratio of friction factor of nanofluid to the base fluid versus Re number

VI. Conclusion

This study presents the experimental investigation of Convective heat transfer performance of Al_2O_3/water nanofluid for 0.5% and 1% volume concentration for laminar flow regime under constant heat flux conditions. It was found that:

1) Convective heat transfer coefficient of Al_2O_3/water nanofluid enhances compared to the base fluid. In addition, increasing particle concentration leads to a better enhancement in convective heat transfer coefficient.
2) The ratio of (h_{nf}/h_{bf}) increases with increasing *Re* number for both volume concentrations. The average heat transfer coefficient enhanced about 11-20% with 0.5% volume concentration and increased about 16-27% with 1% volume concentration compared to distilled water.
3) The Convective heat transfer coefficient enhancement decreases with increasing axial distance.
4) The convective heat transfer coefficient of nanofluid enhances with increase in heat flux. This enhancement is more evident at higher Reynolds numbers.
5) The average ratio of (f_{nf}/f_{bf}) was about 1.10 for 0.5% volume concentration.

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